Heat Pipe Effectiveness Study

Gulf Breeze Laboratory Installation
Pensacola, Florida

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Executive Summary

On October 1, 1996, a Dinh-style heat pipe dehumidification system was installed in the air handling system in Building 49 at the Gulf Breeze laboratory in Pensacola, Florida. The heat pipe was installed to increase the dehumidification capacity of the central cooling system without additional energy consumption. The purpose of this study was to determine the dehumidification effectiveness of the heat pipe installation. The effectiveness was calculated by monitoring temperature and humidity levels of the outside and supply air, and by monitoring the load on the cooling coils and the load on the heat pipe during a two-week period.

A heat pipe dehumidification system is a passive device which uses a series of closed tubes filled with refrigerant to transfer heat from the outside supply air to the supercooled post-cooling coil air. Because the heat pipe pre-cooling coil provides initial air cooling, the total cooling and dehumidification capacity of the system is increased during peak load conditions. During off-peak conditions, the heat pipe pre-cooling coil replaces part of the load on the cooling coil necessary to provide a given level of dehumidification, and the reheat coil replaces part of the reheat load necessary to provide a comfortable supply temperature, with no energy input. As a result, heat pipes provide both enhanced dehumidification and energy savings.

The heat pipe in Building 49 was effective in reducing inside humidity levels about 10%, from an average of 75% before installation to an average of 65% after installation, without affecting the inside temperatures. An additional 20 tons of mechanical cooling would have been necessary to provide this additional dehumidification during peak conditions. The heat pipe cost $42,000 to install, and the additional mechanical cooling necessary to provide the same level of dehumidification would have cost $30,000. Therefore the additional cost of installing a heat pipe instead of mechanical cooling to provide the 10% inside humidity reduction was $12,000.

By recording heat pipe and cooling coil loads for 15-minute intervals during a two-week monitoring period, it was determined that the heat pipe pre-cooling load, and subsequent reheat load, increases linearly with outside temperature. Using a weather bin method analysis, the heat pipe in this location provides a maximum 20 tons of pre-cooling and 240 kBTU/h of reheat with no energy input, saving an estimated 56 kW in peak summer demand, 153,775 kWh in annual energy consumption (about 10% of the total), and $7,700 in annual energy costs. The simple payback of using a heat pipe to provide the enhanced dehumidification for this installation is therefore 15 months. The payback will vary for other installations based on weather data, mechanical system efficiencies, and utility rates. A comparison of the Building 49 utility bills for the 12 months prior to installation and the 12 months following installation, normalized for weather variations, showed an actual energy reduction of 230,750 kWh (14%) and a cost reduction of $9,980.

Introduction

On October 1, 1996, a Dinh-style heat pipe dehumidification system was installed in the air handling system in Building 49 at the Environmental Protection Agency's Gulf Breeze laboratory in Pensacola, Florida. The heat pipe was installed to enhance the dehumidification of the central cooling system without additional energy consumption. The purpose of this study is to determine the dehumidification effectiveness of the heat pipe installation. The effectiveness was calculated by monitoring temperature and humidity levels of the outside and supply air, and by monitoring the load on the cooling coils and the load on the heat pipe during a two-week period.

Background

An energy audit of the Gulf Breeze Laboratory campus, performed by Booz•Allen and Hamilton in March, 1995, identified high humidity problems inside the ductwork and in the laboratory rooms of building 49. Interior humidity levels between 66% and 88% were recorded during a week in June 1994, and occupants had complained of humidity levels near 100% at times. Prior to the heat pipe installation, air was generally supplied to the laboratory at 70 F and 83% relative humidity (RH) for peak summer design conditions. During low-temperature, high-humidity days (such as rainy days), saturated air was being supplied to the laboratory zones, resulting in condensation inside the ductwork. The heat pipe installation was the first phase of a two-stage upgrade to reduce laboratory humidity levels to 50% for peak design conditions. The second phase will involve additional dehumidification capacity.

The heat pipe system was installed on October 1, 1996. The Army Corps of Engineers conducted a study during the summer and fall 1996 - before and after the installation - to determine the effectiveness of the heat pipe system. The result of the study was that the effectiveness of the heat pipe installation could not be determined due to mechanical system shortcomings and insufficient data. This study was conducted during the summer and fall 1997 as another attempt to determine the heat pipe effectiveness, and to address issues raised during the 1996 study.

Heat Pipe System Description

A heat pipe dehumidification system is a set of two coils, one which is placed before the cooling coil (pre-cooling) and one which is placed after the cooling coil (reheat). Closed tubes filled with a refrigerant connect the two coils. The tubing is tilted so that liquid refrigerant settles in the pre-cooling coil. As the outside air passes through the heat pipe pre-cooling coil, it is cooled by the liquid refrigerant. The air then passes through the cooling coil and is further cooled past its dewpoint, and moisture condenses out of the air onto the coil. The air then passes through the heat pipe reheat coil and is heated by the refrigerant vapor to a comfortable supply temperature.

In the heat pipes, the cold refrigerant in the pre-cooling coil is heated past its boiling point by the hot outside air and therefore evaporates. The refrigerant vapor rises up through the heat pipes to the reheat coil. The cold post-cooling coil air passes through the reheat coil, causing the refrigerant vapor to condense, and the liquid refrigerant flows back around to the pre-cooling coil. Since the heat transfer occurs as a result of the alternating evaporation and condensation of the refrigerant, there is no energy required to pre-cool and reheat the air.

Because the heat pipe provides initial air cooling, the cooling coil can drop the post-cooling coil temperature lower than if the pre-cooling were not present. This pre-cooling allows an increased amount of moisture to be removed by condensation at the cooling coil, since less sensible cooling is required to reach the supply air dewpoint. The result is a lowering of the supply air humidity level. During off-peak conditions, the heat pipe pre-cooling coil decreases the load on the cooling coil necessary to provide a given level of dehumidification, thereby reducing energy consumption. This is why heat pipes provide both enhanced dehumidification and energy savings.

Central Air Handling System in Building 49

The central air handling system in Building 49 provides approximately 16,800 cubic feet per minute (cfm) of one-pass air to the laboratory. The system includes a single draw-through fan powered by a 20-horsepower (hp) motor, a filter rack, a steam pre-heat coil, a direct expansion (DX) cooling coil, and a heat pipe dehumidification system. A schematic of the central air handling unit is shown in Exhibit 1. There are two small electric boilers, one which provides steam for the pre-heat coil, and one of which provides steam for air humidification in the winter. According to the on-site maintenance contractor, the steam humidification system has never been used. Fifteen local water-condenser heat pumps provide reheat and space conditioning control for the individual zones.

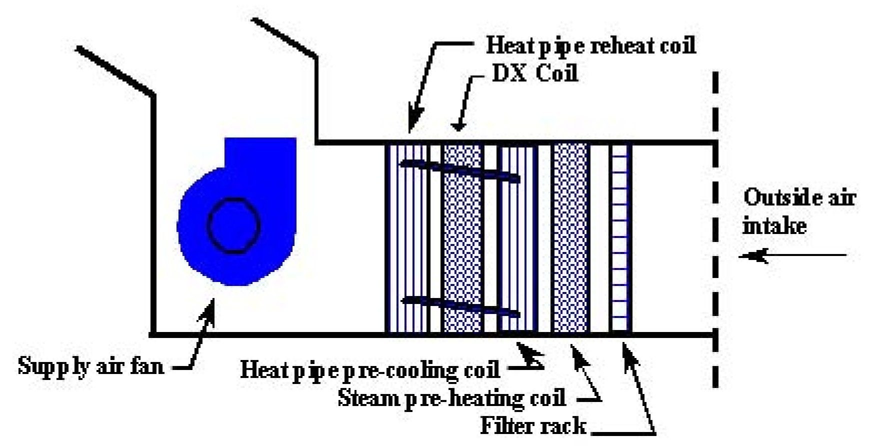

Exhibit 1. Schematic of Building 49 Air Handling System

The cooling coil is divided into an upper half and lower half, with each half containing two refrigeration circuits. There are four compressors, one for each circuit, including three 15-hp compressors and one 10-hp compressor. The total design cooling capacity of the system is 65 tons. A new cooling tower was installed in June 1997, which provides condenser water cooling for the air conditioning refrigeration cycle and the local water-condenser heat pumps.

Before the heat pipe installation, the supply airflow rate was measured to be approximately 19,100 cfm. When the heat pipe system was installed, the additional 0.5 inch static pressure drop due to the presence of two more heat transfer coils in the airstream lowered the airflow rate to approximately 16,800 cfm. The fan was never adjusted to return the airflow rate to its pre-installation level, and since the exhaust airflow did not change, the building is operating at negative air pressure. Returning the airflow rate to 19,100 cfm would result in three changes to system performance:

The higher airflow rate would increase the heat transfer rate between the heat pipe pre-cooling coil and reheat coil, thereby increasing the cooling load and reheat load energy savings for a given outside air temperature. For this installation, it is estimated that the peak heat pipe pre-cooling load would increase about 2.7 tons, which would save an additional 2.7 kW.

During peak ambient conditions when the DX cooling coil is providing its maximum cooling load, the cooling coil cooling and dehumidification will be slightly less than with the lower airflow rate. This is because the load is a function of airflow rate times enthalpy drop, and for a given maximum load, with a higher airflow rate, there will be a smaller enthalpy drop.

Increasing the airflow rate would also increase the energy consumption of the fan system. Using a fan curve analysis, the increase in fan energy consumption to make up for the added 0.5 inch pressure drop would be about 2.0 brake horsepower (bhp), or about 1.5 kW. Based on previous studies and measurements, when the airflow rate is increased the incremental cooling load savings from the additional heat pipe pre-cooling offset the incremental increase in fan motor energy consumption.

The numerical results of this study are based on the lower post-installation airflow rate of 16,800 cfm. If the fan system is adjusted to return the airflow rate to pre-installation levels, further monitoring of the cooling coil and fan motor should be performed to verify energy savings. The other option is to leave the airflow rate at 16,800 cfm and perform a building-wide air balancing and exhaust airflow adjustment to create a positive inside air pressure.

Summary of Army Corps of Engineers 1996 Study

During the summer and fall of 1996, the Army Corps of Engineers monitored the air handling system in Building 49 before and after the heat pipe installation to assess the heat pipe's effectiveness in solving the humidity problem. It developed a report based on the monitoring and concluded that the effectiveness of the heat pipe could not be determined because of problems with the mechanical cooling system, and because pre-installation climate conditions were different than post-installation climate conditions. A review of the Army Corps report revealed the following three factors which are pertinent to this study:

- The cooling tower was undersized, and therefore compressors were tripping off due to high condenser water temperatures during peak cooling conditions. This caused the supply air temperature and humidity to rise during peak conditions. This problem was corrected with the installation of a new cooling tower in June 1997.
- For most of the monitoring period prior to the heat pipe installation (August 17 - September 14, 1996), discharge air temperatures leaving the air handling unit were between 70 F and 75 F and the relative humidity was between 80% and 85%. The high temperature and humidity was due to the frequent failure of DX compressors. The discharge air was further conditioned by the local heat pumps at the individual zones.
- For a seven-day period prior to the heat pipe installation (August 29 - September 4, 1996), the air temperature in room 10 was between 69 F and 73 F, and the relative humidity was between 73% and 80% (average 77.5%). The air temperature in room 4 during this period was between 69 F and 72 F, and the relative humidity was between 68% and 82% (average 73.2%).

Monitoring Plan for this Report

The goal of the monitoring for this study was to determine the amount of cooling that was being provided by the heat pipe pre-cooling coil and the DX cooling coil, and the amount of heating being provided by the heat pipe reheat coil. The supply air energy drop across each of the cooling coils can be calculated by monitoring the air enthalpy difference across each cooling coil and multiplying it with the airflow rate. The sum of the energy reductions provided by the pre-cooling coil and the DX coil is the total cooling provided to the supply air. The energy gain across the reheat coil can be calculated by multiplying the enthalpy gain and the airflow rate.

The temperature and relative humidity of the outside air and supply air was monitored, as well as the temperature of the air between the pre-cooling section of the heat pipe and the cooling coil, and between the cooling coil and the reheat coil. The location of the temperature and humidity sensors can be seen in the duct elevation in Exhibit 2. The airflow rate is 16,800 cfm, as measured by the Army Corps of Engineers after the heat pipe installation.

The cooling coil is divided into an upper and lower half, with the refrigerant flow in each half being provided by two remote compressors. For this reason, it was advantageous to include four air temperature sensors between each section of the heat pipe and the cooling coil, centered in each quadrant, as shown in Exhibit 3.

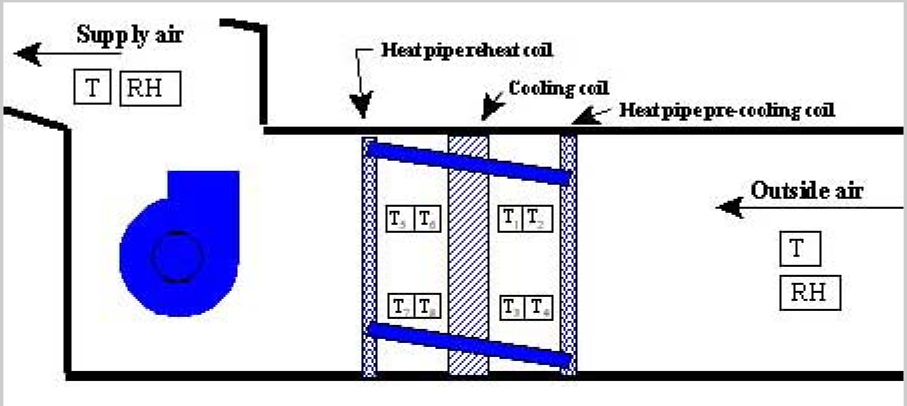

Exhibit 2. Schematic Elevation of Air Handling Unit with Sensor Locations

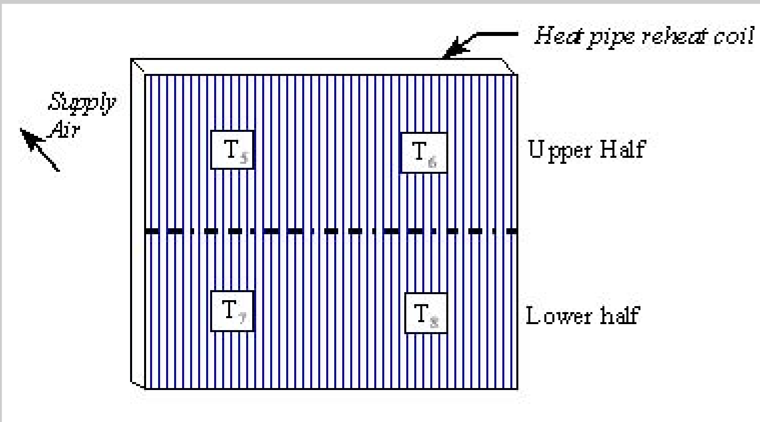

Exhibit 3. Location of Temperature Sensors Between Coils

The energy gain or loss across each coil can be calculated as the product of the enthalpy (heat content) difference and the mass flow of the air. Enthalpy is a function of dry bulb temperature and relative humidity, and can be calculated using psychrometric equations taken from the *1997 American Society of Heating, Refrigeration, and Air-conditioning Engineers (ASHRAE) Fundamentals Handbook* (see Appendix A for the equations and calculations).

By calculating the ambient dewpoint continuously, we can determine if there is any condensation on the pre-cooling section of the heat pipe at any given point in time. If the temperature of the air after the heat pipe pre-cooling coil is lower than the ambient dewpoint, then there will be condensation on the coil. If the temperature of the air after the pre-cooling section is higher than the ambient dewpoint, then there will be no condensation on the coil. The "Cooling Load" spreadsheet (shown in Appendix A) continuously compares the ambient dewpoint with the pre-cooling temperatures, and calculates air enthalpy losses appropriately for when there is condensation or no condensation on the heat pipe pre-cooling coil and DX cooling coil

The "Reheat" spreadsheet (shown in Appendix A) shows the supply air reheat energy provided by the heat pipe reheat coil and fan motor. The heat pipe reheat is the total reheat minus the heat which is added to the airstream by the fan motor. The heat transfer across the reheat section of the heat pipe is entirely sensible, and therefore the post-cooling coil humidity ratio will be the same as the supply air humidity ratio. Using this humidity ratio and the temperature gain, the reheat enthalpy gain can be calculated using the ASHRAE equation for enthalpy shown in Appendix A.

Humidity Levels

In comparison to the measurements taken prior to the heat pipe installation for the same time of year, humidity levels have decreased by about 10%. For outside conditions similar to the pre-installation monitoring period, dry bulb temperatures inside the laboratory were within the same range (between 67 F and 74 F post-installation, between 68 F and 73 F pre-installation), yet average humidity levels have dropped from an average of about 75% pre-installation to an average of about 65% post-installation. Although 65% relative humidity is still high for a laboratory, the 10% improvement can make a difference in occupant comfort and equipment performance levels.

On-site personnel take temperature readings in each area of the building once a day to check the thermostat settings. Two of the more environmentally sensitive rooms, the Mass Spectrometry room and the CM/TC room, are continuously monitored for temperature and humidity using a circular recorder.

The temperature readings show that dry bulb temperatures inside the laboratory rooms ranged from 67 F to 74 F. The readings are consistent with dry bulb temperature readings taken before the heat pipe installation for similar ambient conditions. The circular sheets for the Mass Spectrometry room show that the temperature ranged from 72 F to 75 F, and the humidity ranged from 45% to 68%, between September 12 and 18, 1997. The circular sheets for the CM/TC room for September 15 through 21, 1997 show that the temperature was consistently near 70 F, and the humidity ranged from 60% to 70%.

There still remains a problem with condensation on the ductwork leaking into laboratory rooms through the supply vents. This is due to unconditioned outside air which is infiltrating into the ceiling plenum due to the negative air pressure inside the building. Moisture from this unconditioned air is condensing on ductwork which is transporting cool conditioned air, and is then leaking into the ceiling. This problem can be solved by increasing the airflow rate at the central air handling unit to pre-heat pipe installation levels, thereby creating a slightly positive air pressure inside the building to eliminate infiltration.

Monitoring Results

The results of the temperature and humidity monitoring at Building 49 are shown in Appendix A. For the two-week period between September 11 and September 24, the heat pipe provided an average of 19% savings of cooling system energy, which represents the total amount of additional mechanical cooling which would have been necessary to provide the enhanced level of dehumidification. In addition, the heat pipe provided an average of 4,600,000 Btu daily reheat energy.

Although the monitoring period was not during the peak of the cooling season, there were several occasions when the outside air enthalpy was above the design peak outside air enthalpy (40.5 Btu/lb). Performing a regression analysis to find the heat pipe pre-cooling load as a function of outside air temperature will determine its effectiveness. By comparing heat pipe performance over the range of outside conditions which normally occur during the cooling season, weather bin data can be used to predict heat pipe savings for any location.

The heat pipe was most effective during the peak cooling period of each day. During many days, as the outside air temperature climbed, the load on the cooling coil actually decreased. As is shown by a regression analysis, the heat pipe pre-cooling load increases as the outside air temperature increases, which allows the cooling coil load to drop for a given level of cooling and dehumidification. This is best shown in the cooling load data for September 21, shown in Appendix A and summarized in Exhibit 5. Between 8:00 a.m. and noon, the outside air temperature increased from 79 F to 92 F. The total cooling provided by the system increased from 78.8 tons to 80.7 tons, yet the load on the cooling coil decreased from 67.5 tons to 60.8 tons.

The post cooling-coil temperature is the best indicator of the level of dehumidification being performed. As air is cooled further past the dewpoint, more moisture is condensed on the cooling coil. During the monitoring period, the post-cooling coil temperature was below 60 F more than 86% of the time, with a high reading of 62.8 F and a low reading of 49.1 F.

	Outside Air			HP Pre-cooling	Cooling Coil	HP Pre-cooling	Cooling Coil	Total Cooling
Time	**F**	**RH**	**Dewpoint**	**F**	**F**	**Tons**	**Tons**	**Tons**
8:00 am	79.4	77%	71.9	72.0	57.1	11	67	79
8:15	80.2	74%	71.3	71.8	56.5	13	67	80
8:30	80.7	73%	71.4	72.2	56.9	13	66	79
8:45	81	71%	70.9	71.9	56.7	14	65	79
9:00	81.6	70%	71.1	72.2	56.8	14	66	80
9:15	82.1	67%	70.2	71.8	56.3	16	64	79
9:30	83	64%	70.0	72.2	56.3	16	64	80
9:45	83.7	62%	69.8	72.9	56.1	16	65	81
10:00	84.2	61%	69.8	73.2	56.4	17	64	81
10.15	85.4	60%	70.2	74.4	56.9	17	65	82
10:30	86.3	56%	68.8	74.5	56.4	18	63	80
10:45	87.8	54%	69.4	75.7	57.0	18	64	82
11:00	88.6	53%	69.4	76.1	57.0	19	65	84
11:15	89.4	51%	69.1	77.0	57.7	19	62	81
11:30	90.2	48%	68.3	77.3	57.3	19	61	81
11:45	91.2	47%	68.6	78.3	57.9	20	62	81
12:00	91.8	45%	67.6	78.7	57.4	20	61	81

Exhibit 5. Summary of Cooling Load Data, September 21, 8:00 a.m. to 12:00 noon

Regression Analyses

The effectiveness of the heat pipe increases as the outside air temperature increases. This can be shown by plotting heat pipe pre-cooling load (tons) against outside temperature in an X-Y scatter plot. The load is actually a function of temperature and humidity of the outside air, since the enthalpy of air is a function of temperature and humidity. In order to perform a weather bin analysis, however, it is necessary to give the load as a function of temperature only.

The graph in Exhibit 6 shows heat pipe pre-cooling load in tons, plotted against outside air temperature for the two week monitoring period (September 11 through September 24). Each point represents one 15-minute period. By performing a regression analysis with the data, the heat pipe pre-cooling load, and therefore energy savings, can be predicted using annual weather data. A "best line fit" linear regression, results in the following equation:

Heat Pipe Cooling Load (tons) = 0.527* Outside Air Temperature - 29.8

Standard Error = 2.4 tons

This equation represents the predicted heat pipe pre-cooling load for a recorded outside air temperature. Variations of the predicted load from calculated load are due to variations in outside air humidity.

Exhibit 7 shows the typical yearly weather bin data for temperatures above 65 F for Pensacola, Florida. Using the above equation to calculate heat pipe pre-cooling load, a cooling efficiency of 1.0 kW/ton, and a reheat COP of 2.0 (the averages for the existing mechanical air conditioning system), annual energy savings for this heat pipe installation are expected to be 153,800 kWh ($7,700).

Building 49 Energy Consumption

The heat pipe saves energy in the summer, by replacing mechanical cooling and reheat with "free" pre-cooling and reheat, and does not change energy consumption in the winter. In addition to the heat pipe being installed in the system in October 1996, a new cooling tower was installed in June 1997. However, the effect of the new cooling tower would have been an *increase* in energy consumption, since compressors which had previously failed (and therefore consumed no energy) during peak periods would stay on line.

Exhibit 8 shows monthly electricity consumption and electric demand for Building 49 for fiscal years 1996 and 1997. The chart shows an eight to thirty-five percent reduction in electricity consumption and a thirteen to twenty percent reduction in demand during the summer months (April through September). The one anomaly in the data is the demand for April 1997, when kiloWatt-hour consumption was at the lowest level of the year, yet the peak demand was the highest of the year. It is assumed that there was one unusually hot day in April which drove the peak demand up, yet the weather for the month was mild enough to keep the total consumption low. The table shows a reduction in electricity consumption of 234,260 kWh for fiscal year (FY) 1997 after the heat pipe was installed. The total utility bill reduction from FY 96 to FY 97 was $10,967.

Exhibit 9 shows the monthly consumption data normalized for weather. The adjustments were based on 30-year average cooling degree days for each month where cooling is performed (March through October), and measured cooling degree days for each month in FY 96 and FY 97. Since no cooling is performed November through February, no energy savings during these months can be attributed to the heat pipe. For the cooling months, it is estimated that 230,750 kWh were saved as a result of the heat pipe installation. The utility bill reduction from FY 96 to FY 97 which can be credited to the heat pipe installation was about $9,980.

Conclusion

The installation of the heat pipe in Building 49 has proven effective in dropping inside humidity levels by about 10% while reducing total facility energy consumption by about 230,000 kWh (14%). If the heat pipe were not installed, an additional 20 tons of mechanical cooling capacity would have been necessary to provide the additional cooling and dehumidification. The reheat section of the heat pipe provided additional free reheat which is normally provided by the local heat pumps downstream of the supply fan.

According to the weather bin data energy model, the additional mechanical cooling capacity would have consumed 55,760 kWh ($2,790) per year. Using an average existing heating COP of 2.0, the reheat section of the heat pipe saves an additional 98,020 kWh ($4,900) per year (based on the weather bin data model). The modeled, actual, and weather normalized energy savings are shown below:

Pre-Retrofit Annual Energy Consumption	1,686,260 kWh
Modeled Energy Reduction	153,780 kWh
FY 1997 Energy Reduction	234,260 kWh
Normalized FY 1997 Energy Reduction	230,750 kWh

The total retrofit cost of the heat pipe was approximately $42,000. The installed cost of an additional 20 tons of mechanical cooling capacity is about $30,000, therefore the incremental cost of dehumidification for the heat pipe was $12,000. The modeled annual energy savings is $7,690, therefore the simple payback is 1.6 years. The avoided energy consumption also reduces annual utility greenhouse gas emissions by about 230,660 lb of carbon dioxide, 2,330 lb of sulfur dioxide, and 850 lb nitrous oxides.

Appendix A

Daily Cooling Load and Reheat Load Calculations

The cooling load and reheat load calculations in the spreadsheet are based on psychrometric equations, which can be found in the *American Society of Heating, Refrigeration, and Air-conditioning Engineers (ASHRAE) 1997 Fundamentals Handbook*, and the monitored air temperature and humidity data. The following equations were used for the analysis.

t: temperature (F) **T:** temperature (R) = t + 459.67 **RH:** relative humidity

Water Vapor Saturation Pressure (p_{ws}) = e^X

where $X = C_1/T + C_2 + C_3*T + C_4*T^2 + C_5*T^3 + C_6*\ln(T)$

$C_{1\text{-}6}$: psychrometric constants

Water Vapor Partial Pressure (p_w) = $p_{ws}*RH$

Humidity Ratio (W) = $0.62198*(p_w / p-p_w)$

where p = atmospheric pressure (14.9 psi)

Enthalpy (h) = $0.240*t + W*(1061 + 0.444*t)$

Dewpoint (T_d) = $100.45 + 33.193*\ln(p_w) + 2.319*\ln(p_w)^2 + 0.17074*\ln(p_w)^3 + 1.2063*p_w^{0.1984}$

Cooling load = 4.38*A* h
where A = airflow rate (cfm)
and h = enthalpy drop

Reheat load = 4.38*A* h
where A = airflow rate (cfm)
and h = enthalpy gain

The spreadsheet continuously compares the ambient dewpoint with the pre-cooling temperatures, and calculates air enthalpy appropriately for when there is condensation or no condensation. If the dry bulb temperature of the air after the pre-cooling section of the heat pipe is higher than the ambient dewpoint, then there will be no condensation on the coil. The enthalpy of the air at this point is shown on the spreadsheets as Post HP (NC). The enthalpy drop across the pre-cooling section is a function of the temperature drop, and can be calculated using the following equation:

Enthalpy drop (h) = $[0.24 * (t_o - t_{1\text{-}4})] + W*[1061 + 0.444 * (t_o - t_{1\text{-}4})]$
where h: enthalpy drop (Btu/lb)
and t_o: dry bulb temperature of outside air (F)
and $t_{1\text{-}4}$: average temperature of pre-cooling coil air (F)

If the temperature of the air after the pre-cooling section of the heat pipe is lower than the ambient dewpoint, then some condensation has occurred, and this air will be at saturation. The enthalpy of the air at this point, shown in the spreadsheets as Post HP (C), can be calculated using the ASHRAE equation for enthalpy shown above. The enthalpy drop will then be the outside air enthalpy minus the post-heat pipe enthalpy.